TURBINE ENGINE CONDITION

BY

SONIC ANALYZER METHOD

PRESENTED TO

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AND

MAINTENANCE CONFERENCE

SESSION II
NEW TECHNIQUES ON THE HORIZON

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INTRODUCTION

An analysis of the measurements presently made to insure engine health leads to a natural conclusion. The majority of these measurements are those that may easily be made, i.e., engine temperatures, pressures and speeds. These measurements can give us a rather complete picture of the gas path characteristics of the engine. The more difficult measurements and therefore less often or impossible to make are those related to the mechanical integrity of the engine. While it is true that mechanical deficiencies affect engine performance, more often than not, the performance change noted is an engine stoppage.

Curtiss-Wright has been developing a system capable of detecting incipient failures of engine components with sufficient lead time to failure to schedule component or engine replacement or repair. The system is based upon establishing a relationship between acoustic spectral characteristics and component condition. This work has been supported under contracts from the U.S. Navy BuWeps.

In July of this year, a sonic analyzer developed by Curtiss-Wright began evaluation tests at MCAS, Cherry Point. This paper discusses the basic concept of sonic analysis as developed by Curtiss-Wright as well as some of the results obtained during the analyzer field evaluation and a cooperative program conducted at National Airlines to obtain commercial engine data and experience in Air Line operational procedures.

THE PROBLEM

During an overhaul procedure, each component of the turbine engine is disassembled, checked, and then assembled into the final engine build. The complete engine is then subjected to a post overhaul acceptance test, and if successfully completed, assigned to operational status. A disappointing percentage of these engines become "premature removals". The cause of this infant mortality is most often given as the result of human error introduced during the overhaul phase. While not disputing the cause, the blame should be conditioned by the fact that we have not made available adequate diagnostic tools which would allow the maintenance personnel to discover his errors during the overhaul and postoverhaul acceptance phases.

To examine this point further, let us look at some of the measurements taken during many of the postoverhaul engine checkouts:

1. Thrust
2. Exhaust gas temperature
3. RPM
4. Tail-pipe area
5. Fuel flow
6. Oil pressure
7. Rundown time
8. Vibration (mils, double amplitude)

Measurements 1 through 5 are essentially to trim the engine to performance specifications. Measurement 6 is primarily a damage prevention measurement. If pressures do not meet test specifications, the engine will be shut down. Measurement 7 checks inadequate rotor/shroud clearances and general engine alignment tolerances. Providing these initial measurements meet specification, Measurement 8, vibration, is almost wholly depended upon as an indicator of engine mechanical integrity. It must indicate faults such as:

1. Unbalance
2. Gearbox Condition
3. Bearing Condition
4. Accessory Condition
5. Mechanical Alignment
6. Installation Procedures

If the engine is under the vibration limits, it is accepted regardless of whether the vibration level is 10 or 99 percent of the specified acceptance limits for that particular engine.

There is no question that engine vibration is influenced by each of the above conditions, however, the vibration sensitivities to each of the above conditions varies widely. Figure 1 is a qualitative plot of engine vibration introduced by each of three conditions of varying seriousness. It can be seen that a rather serious bevel gearbox condition will or will not be detected depending on the degree of compressor unbalance which may exist as a bias. It is also evident that for an otherwise healthy engine, the condition of the fuel pump would have to be dangerously close to disintegration before its vibration would be the cause for engine rejection.

The problem of establishing the mechanical condition of the engine components is even more difficult in the field during the engine's operational life since accessibility, equipment and skill levels are less than that obtainable at an overhaul base. What is clearly needed

is a method of establishing confidence in the mechanical integrity of an engine both at the postoverhaul test and during operational use. The Sonic Analyzer as developed by Curtiss-Wright has been demonstrated by field use to be such a device.

CONCEPT FUNDAMENTALS

The sonic analyzer as used is in the same generic family as some new vibration analysis techniques. Both of these techniques start by relating discrete frequencies observed in the spectrograph to individual rotating components of the engine by a knowledge of mechanical details and engine RPM. The choice of microphones vs. vibration transducers was made for several reasons:

1. Installation of vibration transducers in inaccessible locations or their maintenance in hostile environments if permanently installed has always been a problem.

2. The increase in frequency response readily obtained in microphone vs. vibration transducer measurements also showed that the mechanically significant spectrum often exceeds 20,000 cps.

3. The use of non-directional microphones showed that in the case of turbojet engines, very often only two microphone locations were required to detect all mechanically generated discrete frequencies.

Early tests showed that in the analysis of a group of engines almost all rotating component discrete frequencies could be detected. However, for the concept to be useful, it must go further than this. Two requirements must be met:

1. A base line must be established from which deviations in spectral characteristics may be measured.

2. Some mechanism must be found to relate the differences in spectra between base-line engines and non-base-line engines to the condition of the components generating the variable portions of the spectra.

Very early in the program, the fallacy of talking about "average" or "normal" engines was discovered. It was found that many newly overhauled engines contained signals in the sonic spectra that could be related to component condition degradation. It makes no sense to average a slightly degraded main engine bearing with a slightly degraded fuel pump. The base line should not be an "average" engine,

therefore, but the best possible engine or "perfect" engine. In the analysis of approximately 50 engines during postoverhaul acceptance tests, only 4 were classed as base line engines. Figures 2 and 3 compare a section of the spectra from a base line engine and a non-base line engine that were accepted during postoverhaul tests. Note the presence of signals generated by gearboxes and the center main bearing in the non-base line engine.

In the search for mechanically significant spectral changes, tests were conducted to determine if the amplitude of the discrete frequencies could be used. It was found that for some components such as bearings, amplitude could be used. In other cases such as gear trains and pumps, amplitude variations due to load and RPM changes were greater than the mechanically significant variations.

In addition to amplitude, two other signal characteristics were found to give information on the condition of the source. These were signal modulation and pulse shape changes, or harmonic content. Signal modulation may be either amplitude or frequency modulation with some odd spectral characteristics explainable in terms of simultaneous amplitude and frequency modulation.

In amplitude modulation, the carrier frequency is the tone generated by the component, such as gear-mesh frequency. If each tooth meshes in an identical manner, then each mesh will produce an identical sound pulse. The expression for the carrier, simplified to a sinusoidal pulse shape is given by:

$$f(t) = A \sin (2 \pi N \text{ rps } t)$$

where A is the constant pulse amplitude, N the number of teeth and rps is the rotational speed of the gear per second. If some teeth produce different amplitude pulse heights, the variation will be repeated once per gear revolution and the expression becomes:

$$f(t) = A (1 + a \sin \pi 2 \text{ rps } t)(\sin 2 \pi N \text{ rps } t)$$

again assuming sinusoidal wave shape.

The expansion of the expression for the amplitude modulated wave shows that two new frequencies are introduced; N (rps) ± rps. These frequencies are called sidebands and are readily detected in the spectrograph. The ratio of the sideband amplitude to the carrier frequency is a measure of the extent of modulation and therefore the non-uniformity of the gear teeth meshing.

In the example used above, sinusoidal pulse shapes were used. In practice, the pulse shape of the carrier is rarely sinusoidal and this results in harmonics of the fundamental being present. If the pulse shape changes, the relative amplitude of the harmonics change which is readily detected in the spectra produced.

Laboratory rig testing and the careful analysis of turbine-engine spectra have confirmed the usefulness of these two techniques in determining mechanical condition of the engine components. These concepts coupled with the idea of the base-line engine form the basis of the acoustic analyzer as a diagnostic tool.

RESULTS

Curtiss-Wright has produced two models of sonic analyzers and will be delivering the third early in 1966. The first analyzer, the CWEA-1, relied on the skill of the operator to differentiate between base line spectral characteristics and those indicating mechanical discrepancies. This analyzer has been used by Curtiss-Wright personnel to detect mechanical discrepancies in the following components of the J-65 engine:

1. Main Bearings
2. Gearboxes
3. Hydraulic System
4. Accessory Alignment & Installation
5. Compressor Unbalance

The CWEA-2 analyzer has been designed to present a simple go-no-go indication that allows relatively unskilled flight line personnel to use the analyzer. This analyzer was utilized at MCAS Cherry Point by Marine Corp personnel during July 1965. During the one-month period prior to squadron deployment, the following discrepancies were detected using the CWEA-2 analyzer:

1. Center main bearing outer race
2. Compressor unbalance
3. Hydraulic pump

It is interesting to note that the engine exhibiting problems 1 & 2 was operated in the test cell with conventional instrumentation and no indications of problems were evident. The discrepancies were confirmed by tear down and inspection.

The technique used to detect these three problem areas are representative of the concept and are given below:

Center Main Bearing

In a normally operating bearing, random noise is produced. If irregularities are present such as spalling on the races or rollers, discrete frequencies are produced. The frequency may be calculated, depending on whether the spall is on the inner or outer races or on the rolling elements, from the dimensions of the bearing and the shaft rpm. Figure 4 shows the damaged area in the outer race under X5 magnification. Amplitude of the signal produced is used as the indicator of condition. Rotational characteristics of bearings generally produce a spectral rise in amplitude over a span of frequencies rather than a simple discrete. Figure 5 shows the spectral rise centered at the outer-race frequency that is typical of bearing problems.

Hydraulic System

The harmonic content of hydraulic pump signals has been used successfully to detect variation of system component conditions. Figure 6 shows the normal harmonic content of pump signals as detected for an engine installed in an aircraft. Note that only the second and fourth harmonics are present. The lack of odd harmonics is due to double action characteristics of the pump. Figure 7 shows the spectrogram of the hydraulic system signals under a condition of pump seal failure. Note the presence of strong odd harmonics in the spectrogram.

Compressor Unbalance

Each stage of an axial flow compressor produces a discrete frequency of compressor whine which is equal to the number of compressor blades times the rotational speed. The instantaneous amplitude of the compressor signal is dependent on many compressor variables, including blade tip to shroud clearance. In the case of a compressor not perfectly balanced, one would expect to see a variation in the tip clearance depending on the amount of main bearing clearance, i.e., the compressor centerline orbits the engine centerline. Since periodic variation in the tip clearance is expected to give periodic variations in the amplitude of the compressor signal and thus sidebands, a relationship exists between unbalance and the ratio of sideband amplitudes to center frequency. Figure 8 shows the sideband relationship for an engine with out of limits compressor unbalance.

Recordings and analysis have been performed on the following engines: J-65, J-57, J-58, J-79, and JT3D, as well as analysis of power transmission systems. Preliminary work has been done on the JT8D to investigate possible problems in a fully ducted engine. Results of these tests have shown that the engine is treated as a collection of bearings, gear trains, accessories and compression & turbine stages. The ability to treat a gear train as a gear train instead of an integral component of a specific engine allows a large amount of carry over of information from one engine to another. This ability allows the application of the analyzer to new engines or systems without the necessity for extensive and costly component malfunction programs.

To illustrate this point, let us examine the case for bevel gears. In bevel gears, at least one of the gears must be mounted on the shaft end. A common problem in the installation of these gears is the alignment of the shafts so as to produce negligible gear eccentricity. If eccentricity is present, in addition to a small amount of frequency modulation due to changes in the instantaneous speed ratio during one revolution, there is an amplitude modulation due to variations in the mesh of the gear teeth. The spectrogram of the region of the gear-mesh frequency for a J-79 inlet radial gear, Figure 9, clearly shows the presence of sidebands. The normal spectrogram for this region contains only a low amplitude gear-mesh frequency. Note that the displacement of the sidebands is equal in frequency to the rotational speed of the gear.

A cooperative program has been conducted with National Airlines to investigate applicability of the analyzer to commercial engines and in particular, to investigate the effects of the forward fan of the JT3D. The JT3D was picked to allow a comparison with the large amount of J-57 engine data already collected. Based on a limited number of JT3D engines recorded and analyzed, base line engine spectral characteristics have been assigned. Three of the engines analyzed showed large deviations from these base line characteristics. Figures 10 and 11 compare the spectra in the region of 0 to 2000 cps for the base line engine and an engine exhibiting the largest deviations from base line characteristics. Note that all deviations are concentrated in one engine area. The accessory drive gear on the N_2 shaft along with the No. 4 bearing at this location, gear trains and pump signals associated with the fuel system are present.

General

Comparison of the acoustic spectra produced by engines on an inside test stand with those produced by engines installed in aircraft shows that while the overall amplitude of the spectrum is changed, the essential characteristics of the spectra are unaltered. Microphone locations for the test stand installation and the aircraft installation are maintained in the same position relative to the engine. Figure 12 shows the microphone locations for an A4 aircraft.

Spectral analysis of the engine is performed at idle rpm with no mechanical or electrical connections to the engine or aircraft. Data has shown that while some signals are amplified at higher rpm, the spectral characteristics of the signals remain the same. Higher amplitudes are in most cases offset by higher background noise. Analysis at idle rpm allows aircraft check-out on the flight line rather than requiring operation in a high power-run-up area. In the case of suspected problems such as main bearings, idle operation is more desirable both for the safety of the operating personnel and for the engine.

CONCLUSIONS

The acoustical analyzer has proven to be a useful tool in diagnosing mechanical conditions of turbine engines and transmission systems by using techniques such as modulation and pulse-shape analysis. It is capable of treating the engine as a test fixture driving the component under investigation, allowing for rather rapid application of the analyzer to new engines and other rotating systems. Analyzer design is simple; only a small plug in element change is required to convert from engine to engine.

To date, all the Curtiss-Wright effort has been directed towards development of a useful piece of ground support equipment. Tests to date have demonstrated a system capable of detecting incipient failures of engine components with sufficient lead time to failure to schedule component or engine replacement or repair. While experimental investigation of this technique has not been accomplished for airbourne uses, there exists possible applications in this area as well.

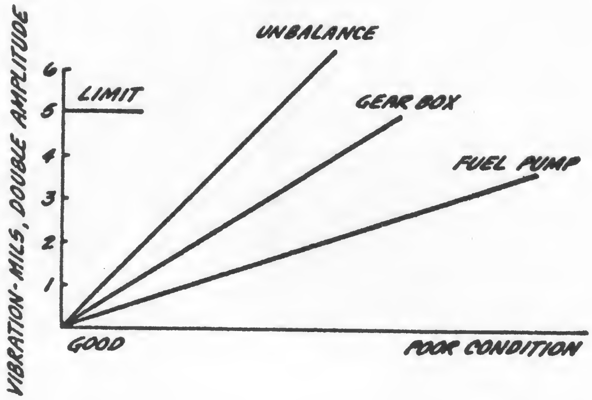

VIBRATION-MILS, DOUBLE AMPLITUDE
6
5
4
3
2
1
LIMIT
UNBALANCE
GEAR BOX
FUEL PUMP
GOOD
POOR CONDITION

FIGURE 1

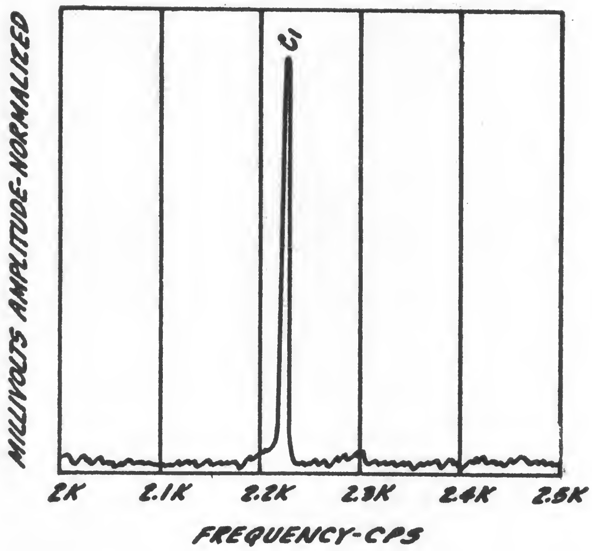
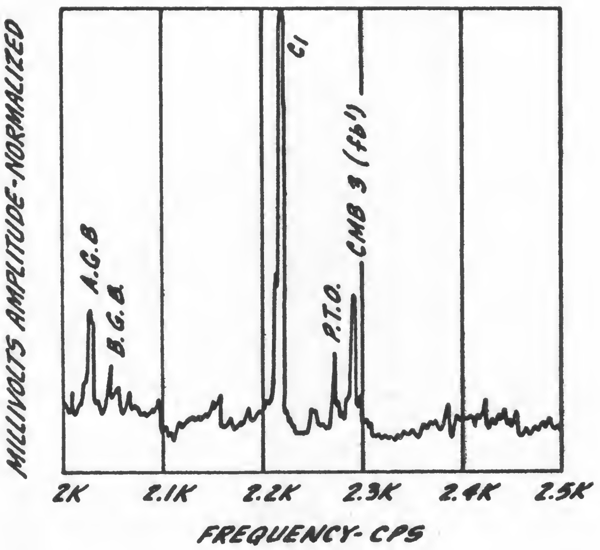

MILLIVOLTS AMPLITUDE-NORMALIZED
C1
2K
2.1K
2.2K
2.3K
2.4K
2.5K
FREQUENCY-CPS

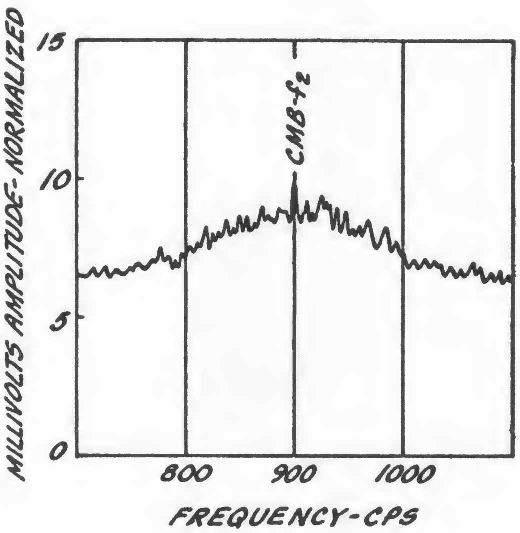
FIGURE 2



MILLIVOLTS AMPLITUDE-NORMALIZED
A.G.B
B.G.B
C1
P.T.O.
CMB 3 (fb')
2K
2.1K
2.2K
2.3K
2.4K
2.5K
FREQUENCY- CPS

FIGURE 3

FIGURE 4



15
10
5
0
CMB-f2
800
900
1000
FREQUENCY-CPS
MILLIVOLTS AMPLITUDE-NORMALIZED

FIGURE 5

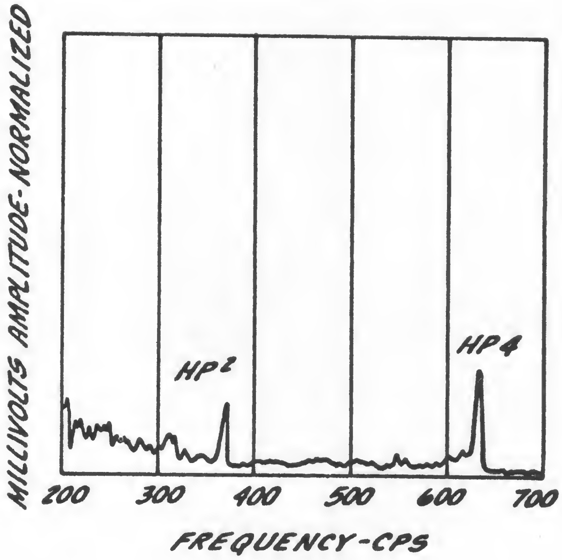

MILLIVOLTS AMPLITUDE-NORMALIZED
HP2
HP4
200
300
400
500
600
700
FREQUENCY-CPS

FIGURE 6

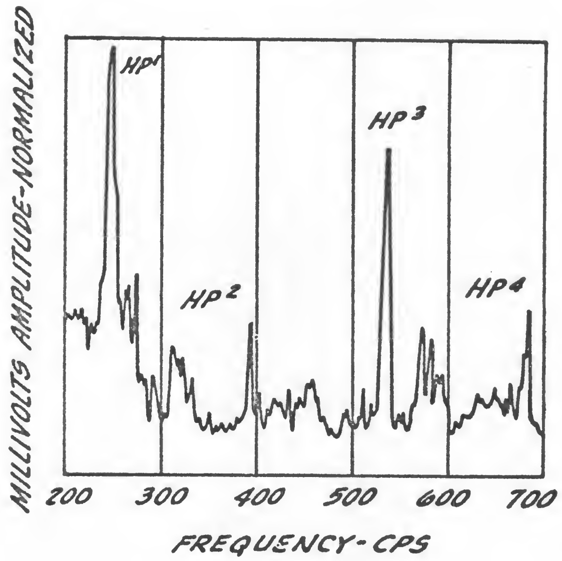

MILLIVOLTS AMPLITUDE-NORMALIZED
HP1
HP2
HP3
HP4
200
300
400
500
600
700
FREQUENCY-CPS

FIGURE 7

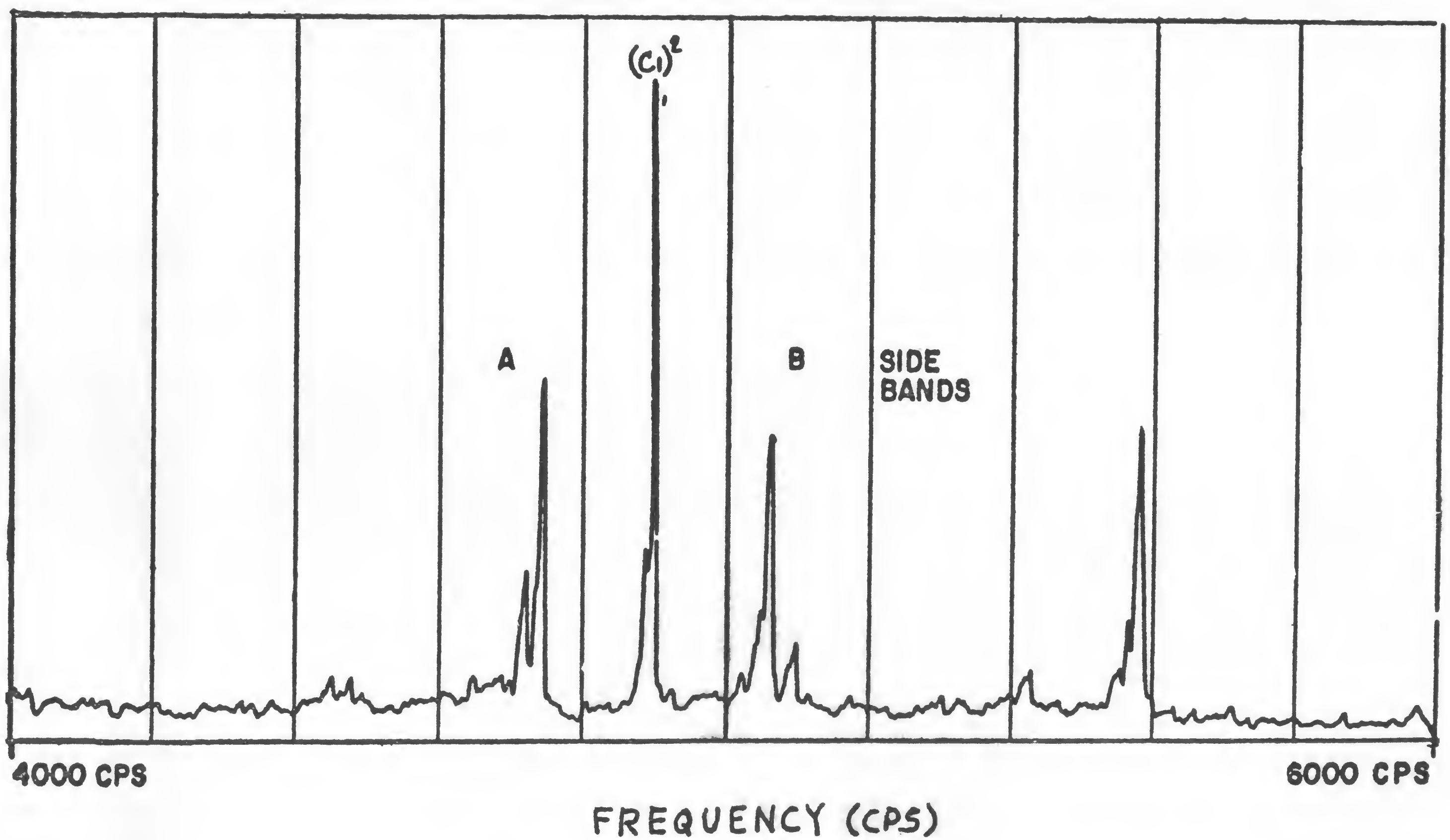


FIGURE 8

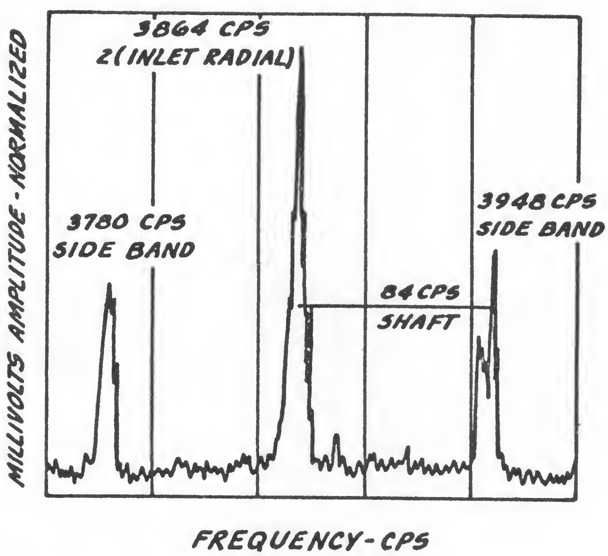

3864 CPS
2(INLET RADIAL)
3780 CPS
SIDE BAND
3948 CPS
SIDE BAND
84 CPS
SHAFT
MILLIVOLTS AMPLITUDE - NORMALIZED
FREQUENCY - CPS

FIGURE 9

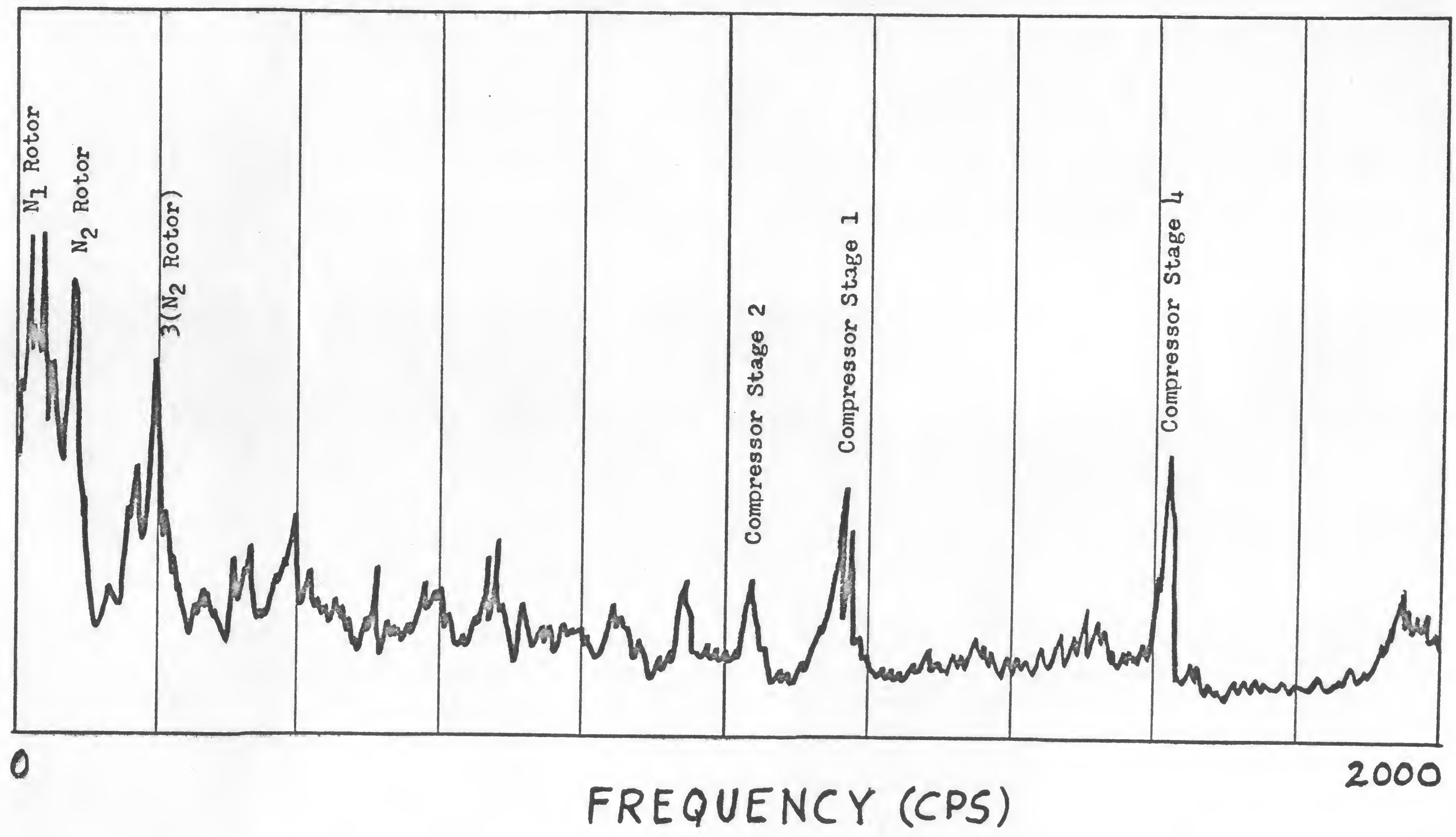
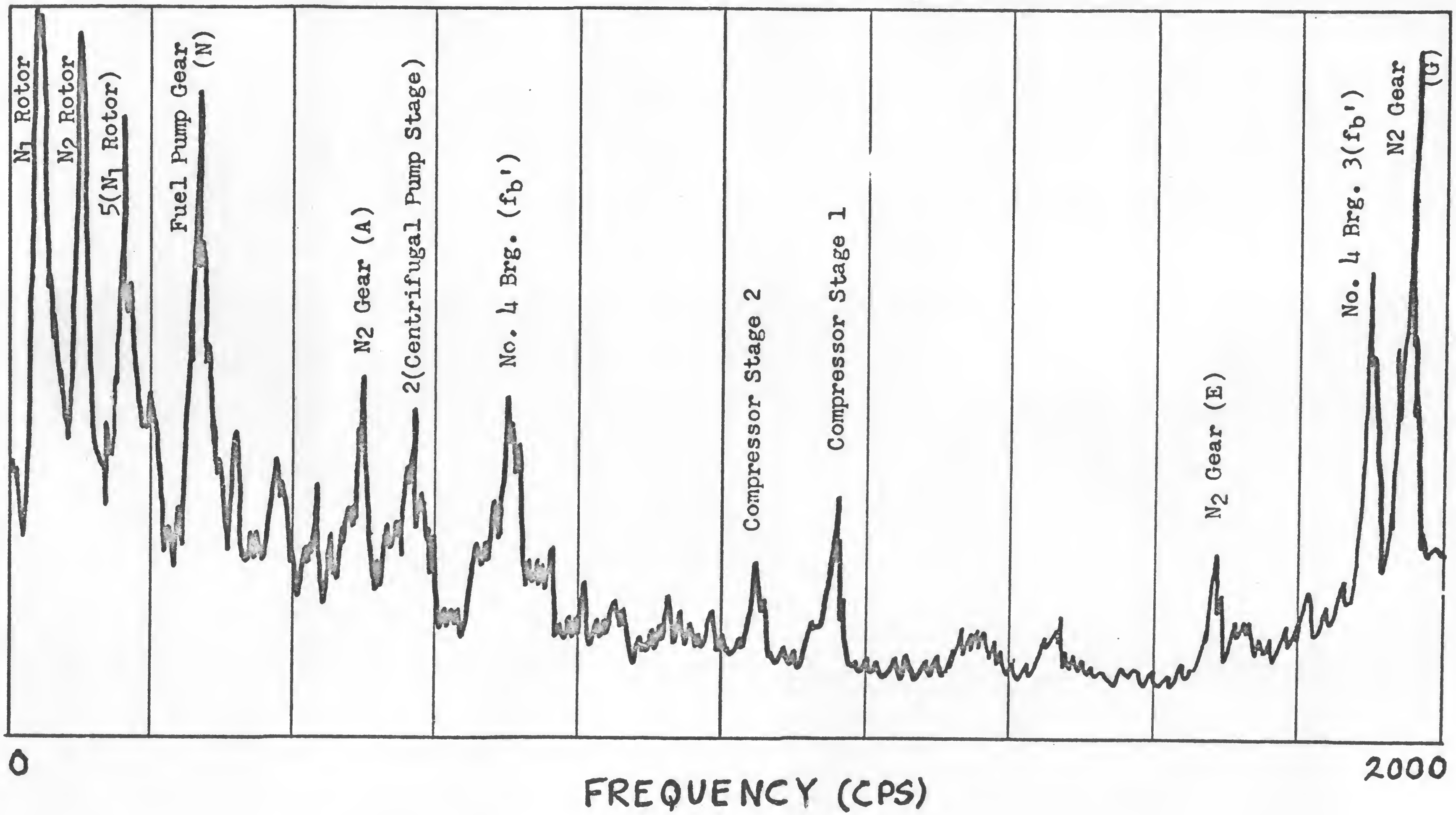

N1 Rotor
N2 Rotor
3(N2 Rotor)
Compressor Stage 2
Compressor Stage 1
Compressor Stage 4
0
2000
FREQUENCY (CPS)

FIGURE 10



N1 Rotor
N2 Rotor
5(N1 Rotor)
Fuel Pump Gear (N)
N2 Gear (A)
2(Centrifugal Pump Stage)
No. 4 Brg. (fb')
Compressor Stage 2
Compressor Stage 1
N2 Gear (E)
No. 4 Brg. 3(fb')
N2 Gear (G)
0
2000
FREQUENCY (CPS)

FIGURE 11

FIGURE 12